Thermal hydraulic comparison of helical coil and bayonet tube steam generators for Small Modular Reactors

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Abstract – *In the present paper a comparative analysis of the thermalhydraulic performance of helical coil and bayonet tube steam generators (SG) is presented. The tool chosen for the analysis is RELAP5-3D/4.0.3. The reference conditions are the ones of the primary and secondary fluids of the SMART Small Modular Reactor. The analysis has been carried out by considering different operating conditions. First, the performance of the SGs in nominal conditions has been compared; subsequently, assuming a power load control at constant average primary temperature, the thermal hydraulic response of the components at different operating conditions has been studied and the helical coil SG results to be the most compact configuration. It's behavior is characterized by higher frictional pressure losses and lower inner heat transfer coefficient if compared with the bayonet tube SG. To analyze the performance of the components out of the nominal condition a region of the map of operation is studied by varying the inlet temperatures of the fluids. A common behavior is found in the considered region. Differences of approximately 12% for the outlet temperatures and 6% for the power have been found.*

I. INTRODUCTION

The Steam Generator (SG) is one of the main components installed in power plants or process plants for energy production. With reference to the nuclear sector, the most common SGs are made of U-tubes or straight pipes. For the future nuclear reactors like Small Modular Reactors (SMR) or generation IV reactors other configurations like spiral, helical or bayonet tube SGs have been proposed. These nonconventional geometries are chosen to meet specific requirements of the new plants: in the case of SMRs, a critical constraint is the compactness of the components to be installed in the primary vessel. Helical or bayonet tube SGs are eligible to be adopted in SMRs thanks to their geometric characteristics; however, it is important to preliminary qualify their behavior in the operating conditions of these kind of plants: in order to do so and to highlight the respective benefits and drawbacks a comparative analysis must be carried out, i.e. the performance of different components designed for the same function has to be compared by considering the specific constraints of a well-defined context. The comparison can be done on the basis of experimental campaigns, numerical simulations, or a combination of the two. In the case of SGs for SMRs the context is the primary system of a nuclear reactor, whose function is to remove thermal power during nominal conditions; from the thermalhydraulic point of view, the performance can be evaluated by means of the values of weight, volume, surface, heat transfer coefficients, heat flux and pressure losses of the components. In the present paper two SGs to be used in an SMR, namely a helical coil and a bayonet tube SG, are considered for a comparative analysis process from the thermal and hydraulic point of view. The reference conditions of the fluids in terms of pressures, temperatures and flow rates and the preliminary design for the helical coil SG are taken from literature on the basis of the SMART SMR reactor[1-4]. The two different SGs are designed to transfer the same amount of power at the nominal conditions of the SMART reactor, ensuring the same flow cross section for the primary coolant. The thermal-hydraulic analysis has been carried out by means of the system code RELAP5-3D/4.0.3. Firstly the performance of the different SGs are compared at full power condition. In the second part of the paper, the comparative analysis is extended to other working conditions different from the nominal ones.

II. THE STEAM GENERATORS

II.A. The helical coil type

Both fluid-dynamics and the heat transfer in the helical geometry have been extensively studied in different fields (energy, chemistry, biomedicine)[5-7]. Several reviews are available in the literature[8-9] collecting the most important scientific results obtained so far. The fluid-dynamics in an helical coil is a complex process, because the inner fluid is subjected to the simultaneous presence of gravitational, inertial and centrifugal forces. The presence of centrifugal forces causes the transition from laminar to turbulent at higher Reynolds numbers, increases the frictional pressure losses and enhances the heat transfer. These effects have been demonstrated by several numerical and experimental studies that compare helical to straight pipes[8,9]. In commercial helical coil SGs the inner fluid enters the bottom region of the bundle in subcooled conditions and exits from the top as superheated steam, whereas the fluid on the outer side flows from the top to the bottom of the bundle. The external flow is influenced by the degree of compactness of the bundle and by the inclination of the tubes. The study of heat transfer between an inner and an outer fluid in an helical coil heat exchanger has been addressed through experimental campaigns, system codes or in-house codes. Esch et al.[10] studied the prediction capability of the system code TRACE for the SG of a gas cooled reactor by implementing different correlations for the Nusselt number and found the best correlation to fit the experimental data. Also Mascari et al.[11] evaluated the prediction capability of the TRACE code in the case of the helical coil SG of the MASLWR test facility in natural circulation. Yang et al.[12] proposed a system code for the steady-state analysis of an helical coil heat exchanger for a SMR reactor and validated it by means of experimental data. The results showed a general agreement except for the steam outlet temperature. Caramello et al.[13] recently proposed a model for the thermal-hydraulic characterization of an helical coil SG during steady state conditions.

TABLE I

Helical coil SG geometrical data

Parameter	Value	Unit measure
Pipes inner diameter	12	mm
Pipes outer diameter	17	mm
Pipes length	24.7	mm
Pipes material	Inconel 690	
Number of pipes	375	
Number of rows	17	
Radial pitch	22.5	mm
Axial pitch	20	mm
Cassette external diameter	1.35	m

The predictive capabilities have been tested against the reference data of the IRIS reactor and the results of the system code RELAP5/Mod.3.3, showing a general agreement except for the dryout region. The geometric characteristics of the present investigation helical coil SG are reported in table I[1-4].

II.B. The bayonet tube type

The most simple bayonet tube configuration consists of two coaxial tubes where the inner tube is open at the bottom to allow flow circulation between them. The feedwater flows in the inner tube from the top to the bottom and then upwards along the annular region between the two pipes; heat is exchanged between the feedwater and the rising fluid in the annular region. Through the external tube the heat is transferred from the fluid inside the bayonet tube to an outer fluid, which in this case is the reactor coolant. The regenerative heat transfer permits to preheat the feedwater that enters the annular region. If high quality steam is required at the SG outlet, the regenerative heat transfer is reduced as much as possible by thermal insulating techniques, such as coatings or paintings, that prevent possible steam condensation phenomena in the annular riser. Nucleation and boiling in the annular region are asymmetric and only occur on the external surface, the only one where temperatures higher than the saturation value can be reached before the dryout region. The annular geometry is particularly favorable for the heat transfer, because it allows higher heat transfer coefficients with respect to the circular cross section.

The first application of these heat exchangers in nuclear industry is found in the first sodium fast reactors[14]. Recently, the bayonet tube configuration has been chosen for the Advanced Lead Cooled Fast Reactor European Demonstrator (ALFRED) as primary SG[15]: each bayonet tube is equipped with a layer of low conductivity paint on the inner tube to produce high quality superheated steam and avoid local condensation, and with a porous matrix between the annular riser channel and the lead to reduce the mixing probability of water in lead and to detect leakages in real time. Dickey et al.[16] studied the heat transfer inside a bayonet tube heat exchanger and created a mathematical model to predict the performance of the heat exchanger both in nominal conditions and reverse flow conditions. Considering the available literature, bayonet tube heat exchangers have been studied less than helical coil heat exchangers. Until now, some open issues are still unresolved regarding the effect of regenerative heat transfer between the inner and outer tube, the effect of the inversion region at the bottom of the bayonet tube and boiling and condensation phenomena in the annular region.

The geometrical characteristics of the present study bayonet tube SG are showed in table II. To provide a suitable insulation between the fluid inside the inner tube and the fluid in the annular gap and to prevent the steam condensation, a 0.5 mm layer of high insulating paint (0.05 W/m/K) in the inner part of the smaller tube is used.

TABLE II

Bayonet tube SG geometrical data

Parameter	Value	Unit measure
Internal pipes inner diameter	10.16	mm
Internal pipes outer diameter	12.7	mm
External pipes inner diameter	15.7	mm
External pipes outer diameter	19.0	mm
Pipes length	3.8	m
Pipe material	Inconel 690	
Pitch	28.5	mm

II. THE NUMERICAL MODELS

The system code RELAP5-3D, which is based on a two-phase nonhomogeneous and nonequilibrium model, has been chosen for the analysis: it is solved by a fast, partially implicit numerical scheme to get economical calculation of the system transients[17]. The partial differential equations governing the fluid-dynamics and heat conduction in the heat structures are solved with a semi-implicit advancement scheme with time step control, by adopting the same time step for heat conduction and hydrodynamics and implicit advancement of the heat conduction/transfer for hydrodynamics.

The same boundary conditions for the hydrodynamic components are adopted to perform the comparative analysis. The boundary conditions are taken from the available data of the SMART SMR reactor[4] and are shown in table III. These boundary conditions are imposed in the models with time dependent junctions representing the inlet flow rates and time dependent volumes representing the outlet pressures.

TABLE III

Boundary conditions

Variable	Value
Primary system inlet temperature [°C]	323.0
Secondary system inlet temperature [°C]	200.0
Primary system flow rate [kg/s]	261.25
Secondary system flow rate [kg/s]	20.1
Primary system outlet pressure [Mpa]	15
Secondary system outlet pressure [Mpa]	5.2

III.A. The helical coil model

Figure 1 shows a sketch of the helical coil SG model that has been implemented in the code.

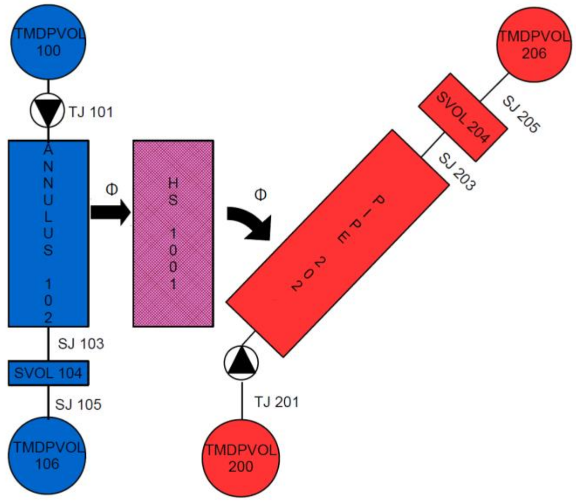

Fig. 1. Helical Coil SG model.

The model of the SG has been developed on the basis of the Idaho national laboratories recommendations[18]. The pipes of the SG are modeled as a single inclined pipe representing the whole bundle (PIPE 202). The length of the pipe is the average length of the helical pipes and the inclination is imposed so as to have the same height difference between inlet and outlet as the real SG. The primary fluid control volume is modeled as an annular component (ANNULUS 102): its flow area is equal to the available flow area for the fluid around the helical coils and the hydraulic diameter is defined on the basis of the flow area and the wet perimeter.

The primary and secondary fluids are set in thermal connection with a heat structure representing the pipes wall (HS 1001). Single volumes are set on the outlets of the active region on both sides (SV 104, SV 204) to monitor the characteristic parameters of the flow such as temperature, pressure and velocity.

Considering the flow conditions of the primary and secondary fluids, specific empirical correlations are required to correctly predict the frictional pressure losses. Ito correlation[19] (equation 1) is used for the friction factor (f) prediction inside the helical pipes in single phase flow, whereas Smith and King correlation[20] has been adopted to evaluate the single phase friction factor for the primary fluid (equations 2 and 3). These equations are implemented in place of classical friction factor correlations contained in RELAP5-3D code by inserting the coefficients in dedicated cards for the definition of the hydrodynamic component.

The friction factor in the helical coil is dependent on the Reynolds number (Re), on the diameter of the coil (D) and on the diameter of the pipe (d). On the other hand, the friction factor on the shell side is dependent on the Reynolds number and on the porosity of the SG (P_y) defined in equation 3.

$$f = 0.07Re^{-0.25} + 0.00725 \left(\frac{D}{d}\right)^{-0.5} \quad (1)$$

$$f = 0.26P_y Re^{-0.117} \quad (2)$$

$$P_y = 1 - \frac{Volume\ of\ the\ pipes}{total\ volume\ available\ for\ the\ SG} \quad (3)$$

The methodology used for the calculation of the frictional pressure losses in RELAP5 for the two-phase flow is based on a two-phase multiplier approach and has not been modified for the calculation.

The models for the calculation of the Nusselt number and consequently for the heat transfer coefficient have not been modified because qualified equations for helical pipes like Mori and Nakayama[21] have not a suitable form to be implemented in the code. Further studies are necessary to determine the weight to be added to the available correlations in the code to take into account the centrifugal forces.

III.B. The bayonet tube model

The bayonet tube SG is modeled as in figure 2. A pipe component is used to model the internal tubes (PIPE 202), while an annulus component simulates the fluids control volume where boiling and superheating occur (ANNULUS 204).

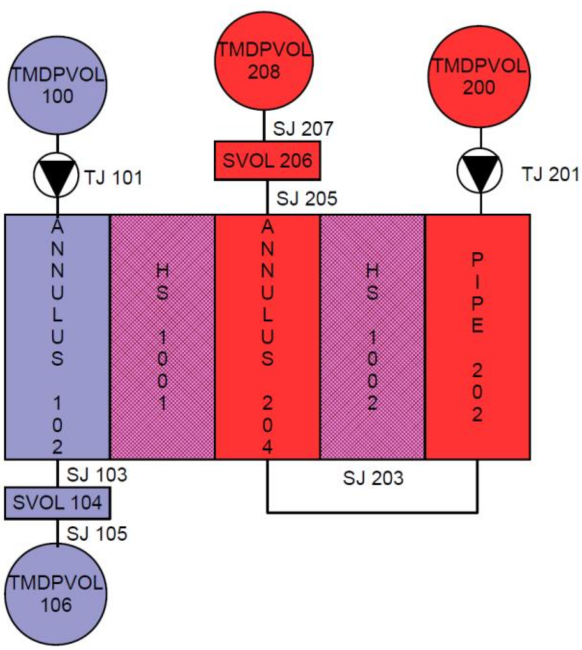

Fig. 2. Bayonet tube SG model.

A multi-layer heat structure (HS 1002), made of Inconel 690 and insulating paint, simulates the internal tube of the bayonet tubes, while the external tube is simulated with the heat structure HS 1001. The external fluid control volume is simulated with an annulus component (ANNULUS 102). Single volumes are set at the outlets of the active region on both sides (SV 104, SV 206) to monitor the characteristic parameters of the flow such as temperature, pressure and velocity. The connection element between the fluid in the inner tube and the fluid in the annular gap (SJ 203) is characterized in order to take into account the localized pressure loss due to flow reversal.

IV. RESULTS AND DISCUSSION

A first comparison of the volumes, the weight and the surfaces of the components can be done on the basis of the design data. Table IV shows the volumes available for the primary and secondary fluid and the required volume of metal. The helical coil SG has a larger volume of secondary fluid (approximately 30% higher). In the bayonet tube configuration, the secondary fluid volume is nearly the same for the inner region and for the outer region. The amount of metal, that is required for the bayonet tube configuration, is approximately three times that of the helical coil SG, which may result in higher fixed costs neglecting the manufacturing process. The whole bayonet tube SG is 40% larger than the helical coil one. The outer heat transfer surface of the bayonet tube SG is 30% lower than that of the helical coil configuration, which results in about the half of the surface density (defined as the ratio between the outer heat transfer surface and the total volume).

TABLE IV

SG volumes and surfaces

Parameter	Helical coil	Bayonet tube
Primary fluid volume [m^3]	2.42	2.42
Secondary fluid volume – inner [m^3]	1.048	0.376
Secondary fluid volume – outer [m^3]		0.381
Secondary fluid volume – total [m^3]	1.048	0.757
Metal volume [m^3]	1.055	3.09
Total volume [m^3]	4.523	6.267
Outer heat transfer surface [m^2]	494.68	340.23
Surface density [m^2/m^3]	109.37	54.29

IV.A. Full power

On the ground of the described models and the boundary conditions that have been shown in table III, table V reports the results of the full power analysis, assuming that the two SGs transfer the same amount of power. The average heat flux of the bayonet tube SG is 45% higher than that of the helical coil SG, which has in turn considerably higher friction pressure losses (see figure

3, which presents the pressure profile along the SGs height for the helical coil SG and for the annular region of the bayonet tube). Larger pressure losses in the helical configuration are due to the combination of a higher mass flow rate, a longer length of the pipes and the presence of centrifugal forces.

TABLE V

Full power results

Parameter	Helical coil	Bayonet tube
Thermal power [MW]	41.25	41.25
Average heat flux [kW/m^2]	83.387	121.24
Mass velocity – inner [$kg/s/m^2$]	473.92	203.34
Mass velocity – outer [$kg/s/m^2$]		200.25
Secondary side pressure loss [kPa]	101.72	18.71
Average global heat transfer coefficient [$W/m^2/K$]	671.1	975.7

The temperature profile of the secondary fluid along the height of the SGs (figure 4) shows a similar distributions of the heat transfer regions, i.e. the non-boiling region, the boiling region and the superheated region. However, even though the SGs heights are the same, the path length of the secondary fluid inside the SG is considerably higher in the case of the helical coil SG, as the fluid flows along a curved line whereas the path in the bayonet tube SG is parallel to the height of the component. The temperature difference at the bottom of the SG between the two cases (8.5 °C) is due to the regenerative heat transfer in the bayonet tube SG. The feedwater flow rate in the bayonet tube SG enters from the top and downflows to the bottom before inverting its direction. In the flow path inside the inner tube the feedwater is preheated by the fluid in the annular region. This phenomenon explains the difference in temperature at the bottom of the SG in the two cases. A difference in the temperature profile can be noticed in the high quality region and in the superheated region. The high quality region, which is characterized by a deteriorated heat transfer, is predicted by the code to be longer in the helical coil configuration.

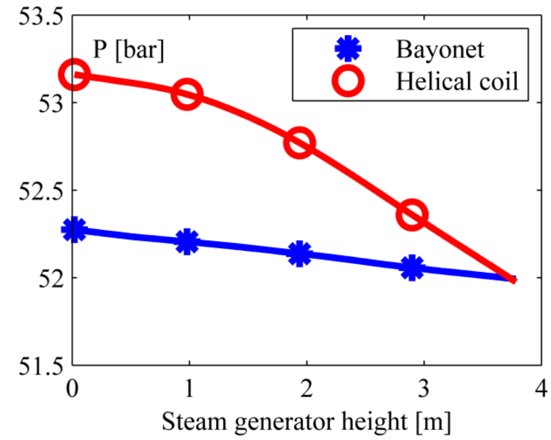

Fig. 3. Pressure profile in the secondary fluid.

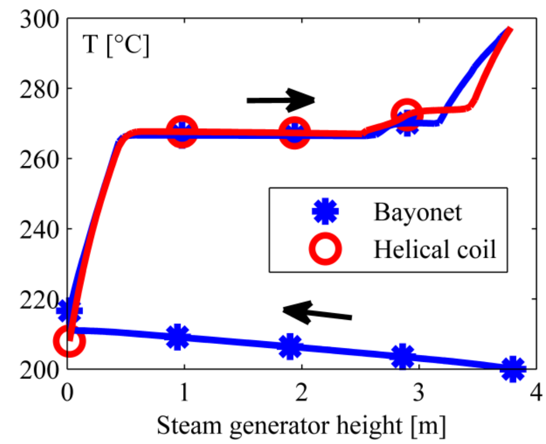

Fig. 4. Temperature profile in the secondary fluid.

The values of heat transfer coefficients are reported in figure 5 for the helical coil SG and for the annular region of the bayonet tube: in the case of the bayonet tube configuration they are generally higher than those of the helical coil configuration, except for the superheated region. The geometrical characteristics of the annular region in the bayonet tube SG corresponds to a lower hydraulic diameter than that of the helical pipe and to a higher Reynolds number, that enhance the heat transfer coefficient. The heat transfer coefficient calculated for the helical coil SG is to be considered slightly underestimated up to a maximum of 15%, as the code is unable to consider the effect of centrifugal forces on the heat transfer. In any case, the heat transfer coefficient in the bayonet tube SG results higher even if the underestimation is taken into account.

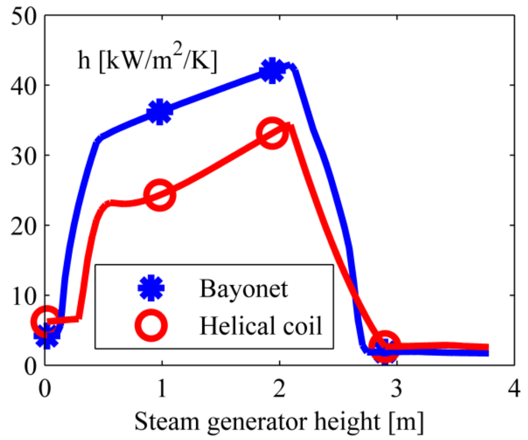

Fig. 5. Heat transfer coefficients profile in the secondary fluid.

IV.B. Behavior at different power loads

The SG must be able to remove heat not only during nominal conditions but also out of the nominal power. In order to remove a certain power, some characteristic parameters of the fluids, such as temperature, flow rates or pressure, are varied by the control system: an example of power control that implies a constant average temperature for the primary system is shown in figure 6. The control logic may also change for different power ranges. Whatever the adopted logic is, the working conditions must belong to the locus of points of the map of operation of the component, which is the locus of points where the component can operate, regardless of the stability.

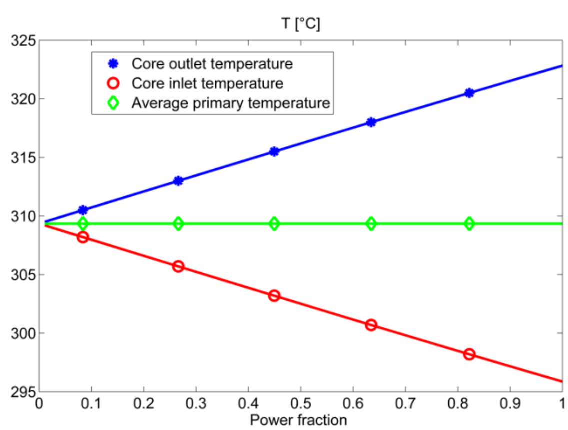

Fig. 6. Constant average primary temperature control.

To obtain the results from the analysis at different loads, a region of the map of operation of the SG has been studied by varying the primary inlet temperature from 300°C to 340°C and the secondary inlet temperature from 160°C to 260°C. Outlet pressures and inlet flow rates are kept constant.

Figures 7, 8 and 9 show some results in the case of the helical coil SG. The data for the bayonet tube are not reported since the general behaviour is qualitatively similar. The variation of the inlet conditions causes a change in the heat transfer regions; in particular, as the inlet temperatures increase the non-boiling length decreases and the superheated length increases. The boiling length is not particularly influenced by the inlet temperature of the secondary system whereas it increases as the inlet temperature of the primary system decreases because a lower heat flux is transferred and therefore a higher surface is required.

The thermal power (fig.7) and the primary fluid outlet temperature (fig.8) show similar trends in the two cases at increasing values of the inlet temperatures of primary and secondary fluids. Maximum variations of the power are of the order of 6% while maximum variations of the outlet temperatures are of the order of 11%. These differences are found in the region where the primary system inlet temperature is lower.

The highest thermal power occurs when the temperature difference between the fluids is maximum (figure 7).

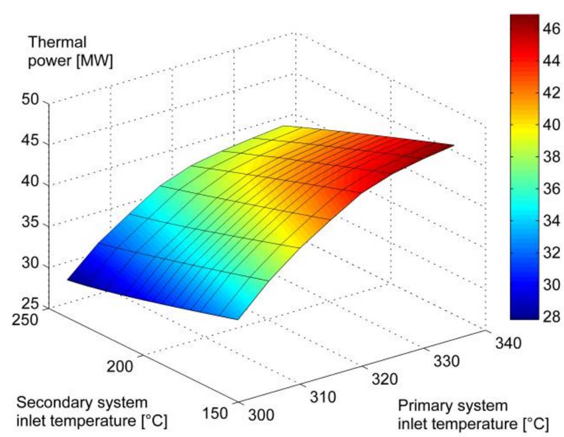

Fig. 7. Power map.

The primary system outlet temperature is influenced mainly by its inlet temperature because the primary system flow rate is about twelve times higher than the secondary system flow rate; therefore a temperature variation in the secondary system has a low impact on the primary side.

At low values of the inlet primary temperature the secondary fluid outlet temperature is constant and equal to the saturation temperature (figure 9), as the secondary fluid is not able to complete evaporation.

Several considerations can be done on the logic that has been used to control the removed power. Assuming a control logic with constant average primary temperature, outlet pressures and flow rates, as the power load decreases the inlet temperature increases and the outlet temperature decreases in the primary side of the SG.

The secondary side temperatures are modified in order to reduce the temperature difference between the fluids and accordingly the inlet temperature must be higher. The maximum value of the inlet temperature of the secondary fluid may be a design limit which will be reasonably lower than the saturation temperature.

In order to control the power with the control logic here hypothesized and to limit the feedwater temperature a minimum value of about 86% of the nominal power can be reached.

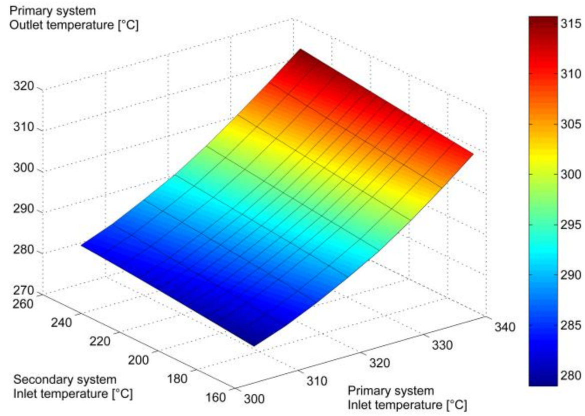

Fig. 8. Primary system outlet temperature map.

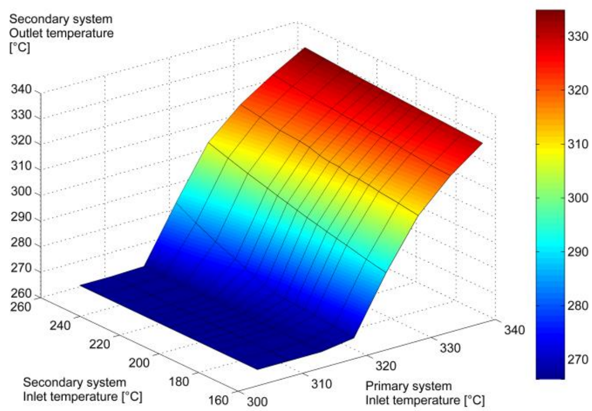

Fig. 9. Secondary system outlet temperature map.

As the power is reduced the nonboiling region reduces in favour of a larger boiling region on the secondary side, and this causes an increase in the friction pressure drop of the order of 15% (figure 10).

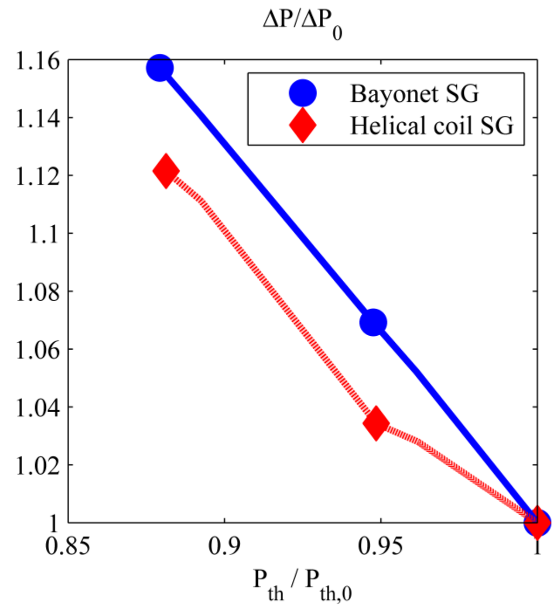

Fig. 10. Pressure drop

V. CONCLUSIONS

The paper presents a comparative analysis of two compact SGs designed to remove the same thermal power and analyses the differences of the components and of their behaviour during nominal conditions and at lower power loads. The analysis shows that the helical coil SG has a more compact configuration concerning the volume of the fluids and the metal amount. Moreover, the helical coil configuration has a higher surface density, which results in a lower heat flux. The bayonet tube SG has significantly lower pressure losses thanks to lower length and mass flow rate and to the absence of centrifugal forces.

The behaviour of the components in nominal conditions is different as regards the values of heat transfer coefficients that are higher for the bayonet tube SG and of the steam temperature in the secondary side.

Finally, the analysis of the behaviour of the components at a lower power load in the specific case of the SMART configuration has highlighted the possibility to use a power control logic with constant average primary temperature down to 86% of the nominal power. The components show a similar behaviour at a lower power load: maximum differences of the order of 6% of power are found by analysing a region of the maps of operation.

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